

Modal Analysis of Compressor Crankshaft



Engineering

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ABSTRACT

The paper presents the analytical and FE modal analysis of a crankshaft. For analytical calculations, the crankshaft is considered as two rotor system to calculate the natural frequency. The three-dimensional finite element model is constructed in PRO-E. The model is meshed in ANSYS workbench. During analysis the six modes are extracted. The results show that the crankshaft is not running in critical speed. The finite element model agrees well with the analytical results and can serve as a baseline model of the crankshaft.

INTRODUCTION

Crankshaft is one of the most important rotating part in reciprocating compressor. It converts the rotary motion into reciprocating displacement of the piston with a four link mechanism. With refer to the researcher every material system containing individual mass and stiffness distribution is susceptible to vibrate [2]. Free vibration analysis is essential to determine the natural frequencies of material system. These are responsible for resonance phenomenon. When the load frequency is matched with one of the natural frequencies the resonance occurs. It leads to high amplitude of vibration.

The calculation of crankshaft vibration performance is difficult because of the complexity of crankshaft structure. Jian Meng explained that the relationship between the frequency and the vibration modal is given by the modal analysis of crankshaft. The results of this provide the base for improvement [1].

3-node Timoshenko beam element method was explained by researcher for the modal analysis of compressor crankshaft [2]. The experimental modal analysis by impact testing method was explained by C. Azoury in his paper [3].

In this paper the compressor crankshafts modal analysis is explained. The Table 1 shows the details of compressor's crankshaft.

Table 1 Crankshaft Details

Crankpin diameter	95 mm
Journal diameter	110 mm
Web width	145 mm
Web thickness	66.5 mm
Flywheel diameter	384.8 mm

The crankshaft is converted in two rotor system analytically and by using Holzer method natural frequency is calculated. This frequency is compared with FEA analysis.

EQUIVALENT LENGTH

A single throw of crankshaft consists of two webs and a crankpin. Depending on the type of the crankshaft, there may be one or two throw between journals. The crankpin usually drives a connecting rod, cross head (for compressor) and piston assembly.

Many equations are given in Ker Wilson for calculating the torsional stiffness of a crankshaft. The basic dimension of the journals, webs, and crankpins are needed, as well as the shear modulus of the shaft material.

The equivalent length of compressor's crankshaft is given by equation 1.

$$L_{eq} = 2B + 0.4(T_1 + T_2) + 3.288 l_{p1} \frac{d_j^4}{d_p^4} + 1.284 R \left(\frac{d_j^4}{T_1 W} + \frac{d_j^4}{T_2 W} \right) \quad (1)$$

Here, T_1 and T_2 are the web thicknesses, W is web width, d_p is crankpin diameter. The equivalent length of crank L_{eq} is calculated by keeping the equivalent diameter of shaft equal to d_j . Substituting appropriate values in equation 1 we get,

$$L_{eq} = 863.26 \text{ mm}$$

The equivalent length of whole crank unit extending from the flywheel to the centre of the crankshaft is 863.26 mm.

MASS POLAR MOMENT OF INERTIA

The polar mass moment of inertia at each throw is depends on the rotating inertia and the reciprocating mass. The total reciprocating mass includes the small end of the connecting rod, cross head, nuts, piston assembly [5]. The mass moment of inertia of reciprocating weight is given by equation (2).

$$I_{Rec} = 0.508 \times \text{Reciprocating Mass} \times R^2 \quad (2)$$

The connecting rod is generally heavier at the crankpin end and lighter at the reciprocating end. The weight distribution of the connecting rod is assumed as two-thirds of the weight is rotating and one-third is reciprocating [5]. The mass moment of inertia due connecting rod is given by equation (3).

$$I_{rot} = \text{connecting rod big end Mass} \times R^2 \quad (3)$$

Since the crankpin rotates about the throw radius and not about its centre, the rotational inertia of journal and crankpin can be calculated by using the parallel axis theorem [5]. The mass moment of inertia of crankpin around crankshaft axis is given by equation (4).

$$I_{cpi} = \left(\left(\frac{\pi}{32} d_{pi}^4 l_{pi} \right) + (8R^2 d_{pi}^2) \right) \times 0.0000072 \quad (4)$$

The inertia of crank web can be calculated by the same way as for crankpin.

$$I_{cwb} = (\text{M. I. of crankweb}) \times T \times 0.0000072 \quad (5)$$

The Table 2 shows the total mass moment of inertias which are calculated from equations 2,3,4,5.

Reciprocating = 40588.52 kg-mm²

Rotating = 14400 kg-mm²

Crankpin = 14349.66 kg-mm²

Crank web = 271991.75 kg-mm²

The total mass moment of inertia of crank throw is given by adding the all inertia determined above.

I1 = 361721.94 kg-mm².

The mass moment of inertia of flywheel is,

I2 = 2046614.96 kg-mm².

The polar moment of inertia of journal of diameter 110 mm is given by equation (6).

$$J = \frac{\pi}{32} d_j^4 = 14373768.14 \text{ mm}^4 \quad (6)$$

NATURAL FREQUENCY OF CRANKSHAFT

For simplicity the total crankshaft system is converted in to equivalent two rotor system as shown in Fig 1.

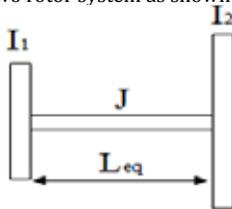


Figure 1 Equivalent Two Rotor System

Nodal point of this system is calculated as N2 = 129.66 mm from flywheel end. The modulus of rigidity of shaft material is 6.86 E10.

The lowest natural frequency of crankshaft can be given by equation 7.

$$f_n = \sqrt{\frac{GJ}{N_2 I_2}}$$

Substituting all values in the equation (7), the natural frequency is, fn = 306.8 Hz.

Holzers Method

Holzer method is a trial and error method and is very effective in finding the natural frequencies of multi rotor system [6].

The calculation by Holzer method is made in XL sheet. The following figure 3 shows the photocopy of XL sheet.

no	rot	Rotor No.	I (kg-m ²)	φ (rad)	θ (rad)	Σθ (rad)	Σφ (rad)	KT	Σ(φ ² /KT)
1840	3316000	1	0.28172	1.22485	1	1.224844	1.22485	134411	0.91132
		2	2.04961	0.92592	0.03888	0.615888	1.74073		
1870	8422900	1	0.36172	1.23790	1	1.237893	1.23789	134411	0.92105
		2	2.04861	0.90454	0.07308	0.551008	1.791		
1900	5610000	1	0.36172	1.25092	1	1.250916	1.25092	134411	0.97182
		2	2.04861	0.74928	0.02849	0.210498	1.46141		
2000	8000000	1	0.36172	1.44039	1	1.440388	1.44039	134411	1.37646
		2	2.04861	0.14848	-0.07568	0.462907	1.9033		
2080	4388100	1	0.36172	1.58004	1	1.580038	1.58004	134411	1.77553
		2	2.04961	0.93932	-0.1735	1.56917	3.14921		
2100	4470000	1	0.36172	1.59519	1	1.595184	1.59519	134411	1.7888
		2	2.04861	0.92597	-0.1568	1.4866	3.08179		

Figure 2 Holzer Method Table

From above Figure 2 of Holzer method table, the lowest natural frequency obtained for this crankshaft is 332.6 Hz.

FEA ANALYSIS

In this section we discuss the modeling of crankshaft, boundary conditions and finite element analysis of crankshaft using FEA. Finite Element method (FEM) simulates a physical parts behavior by dividing the geometry into a number of elements of standard shapes, applying constraints. Uses of proper boundary conditions are very important since they strongly affect the results of the finite element analysis. The drive shaft is modeled in Pro-E. The step file of model is imported in ANSYS workbench.

The main objective of this work is to perform the Finite Element Analysis of crank shaft using CAE Tools, so as to determine the natural frequency in the shaft. The material properties are demanded in CAE to perform analysis. The material used for crankshaft is spheroidal cast iron SG 700/2 and material properties are shown in the table 2.

Table 2 Material Properties

Young's Modulus	176 kN/mm ²
Poisson Ratio	0.275
Density	7.2 kg/dm ³

FINITE ELEMENT MODELLING

Definition of FEM is hidden in the world itself. Finite - any continuous object has finite degree of freedom & it's just not possible to solve in this format. Finite Element Method reduces degree of freedom from infinite to finite with the help of discretization. Element - all the calculations are made at limited number of points known as nodes. Entity joining nodes and forming a specific shape such as quadrilateral or triangular etc. is known as Element. To get value of variable at where between the calculation points, interpolation function is used. Method - There are three methods to solve any engineering problem. Finite Element Analysis belongs to numerical method category.

Finite element modelling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type.

Geometry of Crankshaft

The dimensions of the crankshafts were calculated in above chapters. By using accurate dimensions the solid model of crankshaft is generated by using Pro-E software. The solid model generated of the cast iron crankshaft is shown in Figure 3.



Figure 3 Geometry of Crankshaft

The flywheel is modelled in Pro-E with 384 mm diameter and 134 mm thickness. The assembly of crankshaft and flywheel is made in Pro-E which is going to be used for calculating natural frequency.

Mesh Generation

The geometry is meshed in mechanical model window of an ANSYS 12. The hex dominant method is applied for the geometry. This method is used for applying maximum hexahedron elements to complicated geometry. The body sizing is applied for the whole geometry and element size is given as 5 mm.

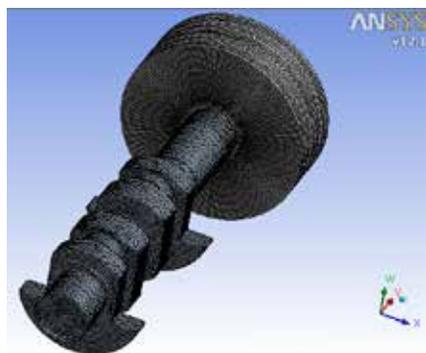


Figure 4 Meshed Geometry

The FE model of the crankshaft geometry is meshed with hexahedral elements, with the global element length of 5 mm and local element length of 0.342 mm at the fillets where the stresses are higher due to stress concentrations. The meshed crankshaft with 484,353 elements is shown in Figure 4.

Boundary Condition

Boundary conditions in the FE model are based on the compressor configuration. The mounting of this specific crankshaft is on three bearings which results in different constraints in the boundary conditions. The three bearing are constrained as simply supported by blocking the displacements at bearing support. At the pump side face of the crankshaft the rotation about crankshaft axis is kept free. The axial displacement is blocked on the pump side face. The analysis is carried out with these boundary conditions.

RESULT AND DISCUSSION:-

The crankshaft can be vibrated at various orientations. The first mode of vibration is the important mode. It is the first node located between flywheel and first crank.

If frequency of any harmonic component of torque is close to the frequency of first mode of vibration, a condition of resonance should be occurred. At this condition the component is said to be running at critical speed.

In the analysis of crankshaft six modes are extracted. The results are tabulated in the table 3. The following figure 5 to 10 shows the six modes of crankshaft.

Table 3 Analysis results

Mode	Frequency in Hz
1	307.22
2	307.28
3	327.99
4	1173.6
5	1229.6
6	1230.8

Figure 5 First Vibration Mode of a Crankshaft

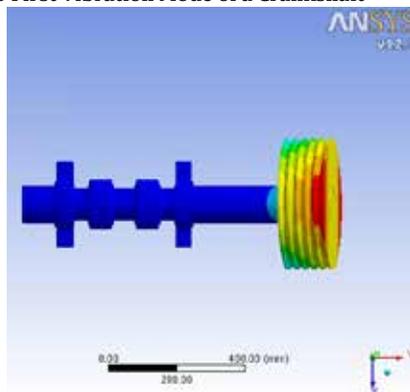


Figure 6 Second Vibration Mode of a Crankshaft

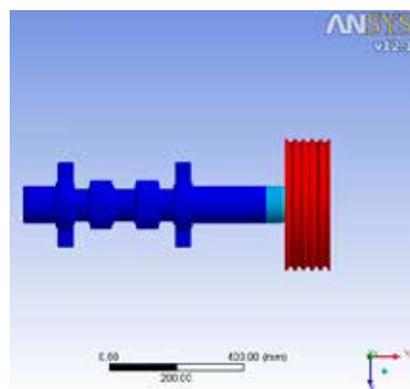


Figure 7 Third Vibration Mode of a Crankshaft

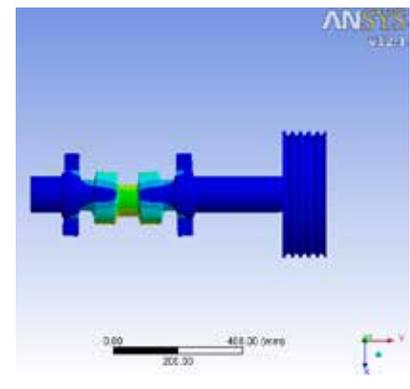


Figure 8 Forth Vibration Mode of a Crankshaft

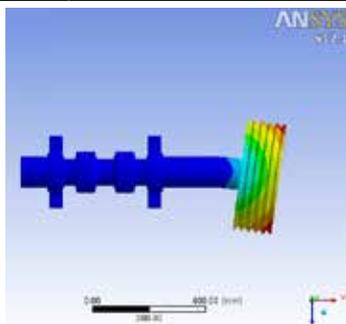
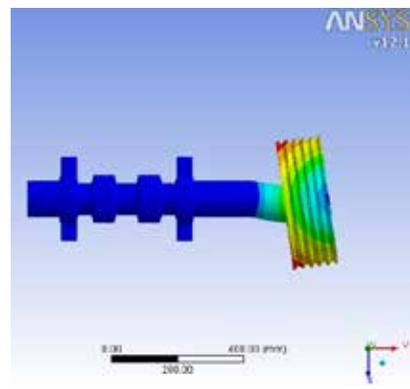
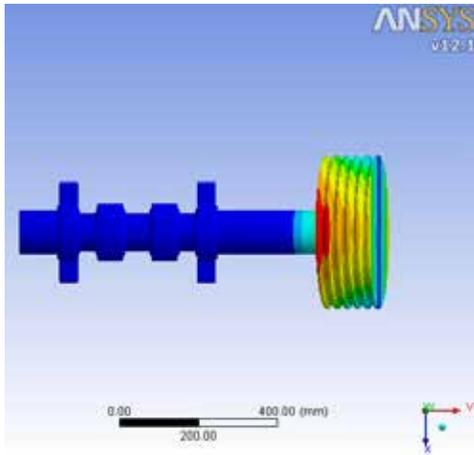


Figure 9 Fifth Vibration Mode of a Crankshaft**Figure 10 Sixth Vibration Mode of a Crankshaft**

The difference between first frequency by theoretical method and by ANSYS is about 7.6%.

CONCLUSION

In this paper the analytical method for calculation of natural frequency of crankshaft is described. For that the crankshaft is converted into two rotor system and natural frequency is determined by Holzer method. This frequency is lowest natural frequency of crankshaft.

The crankshaft is modelled in PRO-E and analysed in ANSYS for its natural frequency. Six modes are extracted during modal analysis. The difference between first frequency by theoretical method and by ANSYS is about 7.6%.

Forcing frequency of crankshaft is 24.91 Hz is very less than natural frequency of crankshaft. So, condition of resonance is avoided and crankshaft is not running in critical speed.

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